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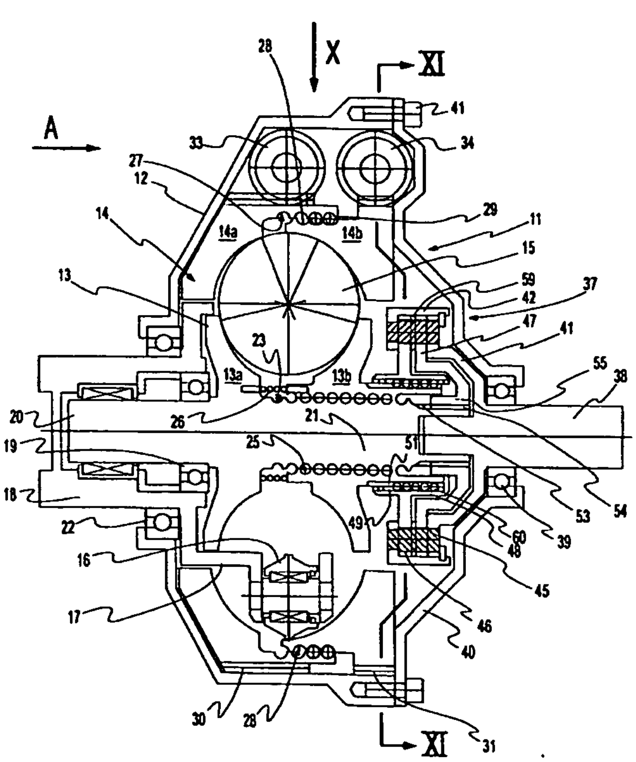


(57) Abstract: A continuously variable drive transmission (11) for transmitting torque between input and output drive members (18, 38), of the type having planetary members (15) in rolling contact with radially inner and outer races (13, 14) each comprising two relatively axially movable parts (13a, 13b, 14a, 14b) for selectively the radial position of the planetary members (15) in rolling contact with them, and means (23, 24, 25) for controlling the axial separation of the two parts (13a, 13b) of the other race (13) to determine the transmission ratio between the input and output drive members (18, 38), in which there are provided means (44-59) for establishing a driving connection between one of the said input and output drive members (18, 38) and one or other of the said two parts (13a, 13b) of the said other race (13) in dependence on the direction of the torque applied between them whereby to transmit torque between the said two drive members (18, 39) in either directional sense. Means are also provided for limiting the maximum radial excursion of the planetary members (15) in one radial direction independently of the separation of the said two parts (14a, 14b) of the said one race (14), whereby to determine the transmission ratio of the device at one end of its range of adjustment.

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# A CONTINUOUSLY VARIABLE DRIVE TRANSMISSION DEVICE

The present invention relates generally to a continuously variable drive transmission device, and particularly to a drive transmission device in which forces are transmitted by rolling traction.

In the applicant's earlier Patent Application No. PCT/GB99/00075 there is described a continuously variable drive transmission for transmitting drive between an input and an output drive member and having radially inner and outer races separated into axially spaced two parts which are relatively axially displaceable and between which are located planetary members the radial position of which is determined by the relative separation of the two parts of one of the races. The other of the races is provided with a mechanism which allows the two parts to move apart or together in order to accommodate changes in the radial position of the planetary members consequent on an adjustment in the axial separation of the two parts of the said one race. This mechanism is effectively torque sensitive in that the forces exchanged between the contacting surfaces of the races and the planetary members is varied in dependence on the torque applied between the input and

output drive members.

This is achieved by means of a helical interengagement of the two members of the said "other" race in the form of a screw thread. As torque is applied the two parts of the race are caused to tend to turn in relation to one another in a sense such that the screw threaded engagement between them causes relative approach thereby increasing the contact forces between the races and the planetary members. Correspondingly, if the torque is decreased, for a given direction of rotation, the forces on the contacting surfaces between the two parts of the "other" race and the planetary members decreases and this continues to the point where, when there is no torque, the contacting forces fall substantially to zero.

It will be appreciated that in the following description references to torque and relative direction of torque relate to the rotational forces applied between input and output members and that these are independent of the speed and/or the absolute direction of rotation of each of the drive members. That is, the torque applied between the input drive member and the output drive member may be in one direction or the other even for a given absolute direction of rotation of the input drive

shaft and the output drive shaft as a consequence of changes in the circumstances of the machinery driven by the transmission device. For example, in a motor vehicle, for a given direction of rotation of the input drive shaft and the output drive shaft (for example in forward motion of the vehicle) the torque may be transmitted in one directional sense when the vehicle is accelerating but in the opposite directional sense when the vehicle is decelerating in so called "engine braking" conditions.

In such a situation it is of value for the drive transmission device to be able to transmit torque in each of the two directional senses as this will allow the above mentioned engine braking effect to contribute to the decelerating forces on the vehicle when the engine output is reduced, allowing greater control of the vehicle's motion without being entirely reliant on wheel brakes.

However, in a structure in which the relative forces between the two parts of the said other race and the planetary members reduces to zero as the torque falls it is necessary to seek some means by which the torque between input and output members can be increased

regardless of the direction of relative rotation of the two drive members.

This is achieved, according to the invention, by a continuously variable drive transmission for transmitting torque between input and output drive members thereof, of the type having planetary members in rolling contact with radially inner and outer races each comprising two relatively axially movable parts, with control means for selectively varying the axial separation of the two parts of one race and thus the radial position of the planetary members in rolling contact therewith, and means for allowing the axial separation of the two parts of the other race to vary whereby to determine the transmission ratio between the said input and output drive members, in which the two parts of the said other race are helically interengaged and there are provided means for establishing a driving connection between one of the said input and output drive members and one or other of the said two parts of the said other race in dependence on the direction of the torque applied between them whereby to transmit torque between the said two drive members in either directional sense.

The present invention also comprehends a continuously

variable drive transmission for transmitting torque between input and output drive members thereof, of the type having planetary members in rolling contact with radially inner and outer races each comprising two relatively axially movable parts, with control means for selectively varying the axial separation of the two parts of one race and thus the radial position of the planetary members in rolling contact therewith, and means for allowing the axial separation of the two parts of the other race to vary whereby to determine the transmission ratio between the said input and output drive members, in which there are provided means for interconnecting the two parts of the said other race with one of the said input and output drive members in such a way that torque is transmitted in either directional sense between the said input and output drive members.

Preferably the two parts of the said other race are helically interengaged with one another and the drive connection to the said other race includes a lost motion coupling.

In a preferred embodiment of the invention the drive connection to the two parts of the said other race comprise two drive transmission members connected to

respective parts of the said other race by a coupling which is sensitive to the direction of relative torque between the drive members.

In embodiments in which there is a lost motion coupling in the drive connection this preferably has a maximum range greater than the range of movement of the drive connection involved in the greatest ratio change from maximum to minimum of the drive transmission.

The means for interconnecting the two parts of the said other race with one of the said input and output drive members may comprise two rotatable drive transmission members mounted for rotation with respective parts of the said other race and driven by axially extending members passing through arcuate slots therein.

Such drive transmission members preferably comprise discs mounted in a housing and the dimensions of which are such as to be a sliding fit therein whereby to define a fluid damper chamber between the said arcuate slots and the drive transmission members.

Preferably the said two parts of the said one race are also helically interengaged such that relative rotation

thereof causes relative axial displacement, and the relative rotation of these two parts may be controlled by individual prime movers such as respective electric motors. In a preferred embodiment of the invention the drive motors for adjusting the relative rotation of the two parts of the said one race are linked by a worm and wheel coupling to each part.

Apart from the ability to transmit drive in either directional sense it is also important for a drive transmission device to be capable of transmitting drive with a minimum of transmission losses, especially when transmitting drive in the highest ratio. It is known that road vehicles consume the majority of fuel when top gear is selected and for some patterns of use, particularly where a large amount of motorway driving is concerned, the consumption of fuel when the vehicle is in top gear may account for 90% or more of the total fuel used by the vehicle. Under such conditions maximum transmission efficiency is of utmost importance in order to improve fuel consumption.

A rolling contact drive transmission device is always subject to internal losses whilst the planetary members are rolling in relation to the inner and outer races due

to the nature of the rolling contact between the planetary members and the races which involves a degree of "spin" at the contact patch. The present invention seeks to provide a drive transmission device in which, in the highest gear ratio selectable, the planetary members will not be required to rotate with respect to the races whilst nevertheless transmitting torque between them.

According to this aspect of the present invention, therefore, a continuously variable drive transmission device of the type having planetary members in rolling contact with radially inner and outer races each associated with respective drive input and output members and each comprising two axially spaced parts, with control means for selectively varying the axial separation of the two parts of one race and thus the radial position of the planetary members in rolling contact therewith, and means sensitive to the torque between the said drive input and output members whereby to determine the separation of the two parts of the other race and thus the transmission ratio of the device, is provided with means for limiting for maximum radial excursion of the planetary members in one radial direction independently of the separation of the said two parts of the said one race, whereby to determine the

transmission ratio of the device at one end of its range of adjustment.

Naturally, for drive transmission of a motor vehicle, the end of the range of adjustment which is most suitable to provide this feature will be that corresponding to the highest transmission ratio. This may be considered to be equivalent to the so-called "direct drive" achieved by a conventional gearbox. By arranging for the torque transmission to take place without relative rolling contact between the planetary members and the races the losses involved upon such rolling contact are avoided.

In a preferred embodiment of the invention the drive input to the device is applied via the radially inner race and the drive output taken from followers associated with planetary members, the radially outer race being fixed against rotation. Preferably, the planetary members are intercalated with follower members carried on a common guide by which drive transmission from the planetary members to the output member is effected. In such a configuration the common guide may include means for limiting the radially outward excursion of the planetary members.

Such radial excursion limiting means may comprise a part of a cage within which the planetary members are constrained to move, or may be provided by an annular member located at or adjacent the common plane of the centres of rotation of the planetary members and encircling these latter.

The surfaces of the radially inner race for contacting the planetary members, preferably extend radially outwardly to a point such that the contact between the race and planetary members takes place at surfaces of the planetary members substantially normal to the axis of rotation thereof.

Embodiments of the present invention will now be more particularly described, by way of example, with reference to the accompanying drawings, in which:

Figure 1 is an axial sectional view of a first embodiment of the present invention in a configuration resulting in a high transmission ratio;

Figure 2 is an axial sectional view of the embodiment of Figure 1 shown in a configuration resulting in a low transmission ratio;

Figures 3a to 3d are schematic diagrams illustrating the relative positions of components of the embodiment of

Figure 1 when clockwise and anticlockwise relative torque is applied between the input and output drive members in a high transmission ratio;

Figures 4a to 4d are corresponding schematic diagrams illustrating the corresponding positions of the same components as are illustrated in Figure 3, when the drive transmission device is in a configuration resulting in a low transmission ratio;

Figure 5 is a schematic diagram, partly cut away, seen in the direction of the arrow X of Figure 1;

Figure 6 is a schematic sectional view taken on the line VI-VI of Figure 1;

Figure 7 is a schematic axial sectional view of an alternative embodiment of the present invention; and

Figure 8 is a schematic end view of the embodiment of Figure 6.

Referring now to the drawings, and particularly to Figure 1 thereof, a rolling traction drive transmission device generally indicated 11 comprises a casing 12 housing radially inner and outer races 13, 14 axially separated into two parts 13a, 13b and 14a, 14b. Between the races 13, 14 are located planet balls 15 between which are located respective planet follower rollers 16 mounted on a follower cage 17 having a tubular axial extension 18

which constitutes one of the drive members of the device. The planets and/or the races may be made of any material capable of supporting the loads exerted on them in use of the device, and in particular may be a ceramic material which has the benefit of having a high modulus of elasticity which gives it a high rigidity. The contact patches are therefore small and efficiency is high.

The two axially separated parts 13a, 13b of the radially inner race 13 comprise a first part 13a having oppositely directed axially extending shafts 20, 21. The shaft 20 is borne on the casing 12 by two bearings 19, 22 the former between the shaft 20 and the tubular drive member 18 and the latter between the tubular member 18 and the casing 12. The other shaft 21 carries a set of helical grooves 23 housing balls 24 which also run in corresponding helical grooves 25 in the second part 13b of the radially inner race 13 to constitute a helical interengagement or threaded coupling between the two parts 13a, 13b allowing them to rotate with respect to one another with low friction provided by the balls 24, and at the same time vary their relative axial positions. A light torsion spring 26 pre-loads the two parts 13a, 13b of the inner race 13 towards one another to maintain initial contact. As an alternative, where heavy loads

are involved, the balls 24 may be replaced by rollers, corresponding changes being made to the shape of the grooves.

Correspondingly, the two parts 14a, 14b of the radially outer race 14 are likewise interengaged by a helical interengagement or threaded coupling comprising helical grooves 27 in the first part 14a of the outer race 14 and corresponding helical grooves 28 in the second part 14b, the cross-section of these grooves being substantially semi-circular whereby to house balls 29 providing a low friction coupling between them.

The radially outer surfaces of the two parts 14a, 14b of the radially outer race 14 have axially extending radial teeth 30, 31 engaged by respective worm wheels 32, 33 driven to rotate by respective electric motors 34, 35. As can be seen in Figure 5, the worm wheels 33, 34 and correspondingly the motors 35, 36 are inclined to the plane normal to the axis of rotation of the device such that the line of contact between the worm wheel 33, 34 and the pinion teeth 30, 31 is parallel to this axis. By driving the electric motors 35, 36 in one direction or the other the worm wheel connection causes the two parts 14a, 14b of the radially outer race 14 to rotate with

respect to one another thereby causing them to approach or separate, depending on the direction of relative rotation, thus causing the planetary balls 15 to adopt a radial position which depends on the relative separation of the two parts 14a, 14b. In this respect it will be appreciated that the curved surfaces of the race 14 contacting the planetary balls 15 have a radius of curvature which is somewhat greater than the radius of curvature of the planets. Likewise, the radius of curvature of the surfaces of the radially inner race 13 which contact the planetary balls 15 are also of a greater radius of curvature than that of the balls 15.

Because the load-bearing ability of materials such as ceramics is so high the efficiency of the device may be increased by providing annular tracks or " flats" (strictly, conical surfaces) around the balls to provide a plurality of incremental ratios. Indeed such tracks may be convex in cross section in order to minimise the size of the contact patch and therefore minimise the frictional resistance to motion between the balls and the tracks.

In Figure 1 the two radially outer parts 14a, 14b of the outer race 14 are shown at a maximum separation allowing

the planetary balls 15 to adopt their furthermost radially outer position, corresponding to which the two parts 13a, 13b of the radially inner race 13 are in their position of closest approach. In this embodiment this defines a high transmission ratio as will be appreciated from a consideration of the relative radial positions of the points of contact between the planetary members 15 and the inner and outer races 13, 14 bearing in mind that the radially outer races are held stationary with respect to the casing 12 and drive transmission takes place between the radially inner race 13 and the planet follower cage 17. In other embodiments (not shown) the drive may be applied to a different member, for example drive may be transmitted from the shaft 18 to the shaft 38 in which case the configuration shown in Figure 1 would be a low transmission ratio. In general, it should be stated, the described structure could be used by restraining any one of the three major components, namely the inner race, the outer race and the planet follower carrier, applying torque between the other two by driving one or the other. In this embodiment it has been chosen to restrain the outer race.

The radially inner race 13 is connected via a drive coupling generally indicated 37 to a drive shaft 38 borne

by bearings 39 on an end plate 40 secured by bolts 31 to the casing 12. The drive shaft 38 has a radial flange 41, which forms part of the lost motion drive coupling 37, which will be described in more detail below.

The lost motion drive coupling 37 comprises an outer cylindrical sleeve 42 with a radial flange 43 having two diametrically opposite holes housing one end of each of two drive pins 44, 45 the other ends of which are received in corresponding diametrically opposite holes formed in the flange 41 of the drive member 38.

Between the flange 41 of the drive member 38 and the radial flange 43 of the cylindrical sleeve 42 are located two discs 46, 47 a first of which, the disc 46, has a radially inner axial sleeve 48 with axial splines 49 housing balls 50 engaged in corresponding axial splines 51 in a tubular extension 52 of the second part 13b of the radially inner race 13. The second shaft 21 of the first part 13a of the radially inner race 13 has a set of splines 53 at its right hand end as viewed in Figure 1, which are engaged by corresponding splines 54 on a tubular boss 55 of the disc 47. The disc 47 is thus fixed for rotation with the first part 13a of the radially inner race 13 whilst the disc 46 is fixed for

rotation with the second part 13b of the radially inner race 13. The second part 13b, however, can move axially with respect to the disc 46 by virtue of the splines 49, 51 interengaged by the balls 50. By contrast the splined coupling 53, 54 between the first part 13a of the race 13 and the axial boss 55 of the disc 47 is axially fixed.

In Figures 3 and 4 the discs 46, 47 are shown schematically in that the notional section line passes through the median plane of the disc rather than constituting a plane perpendicular to the axis. The disc 47, therefore, is represented in Figures 3 and 4 as a flat disc with a splined coupling to a solid shaft whereas the disc 46 is illustrated with both the balls 50 forming the axial splined coupling between the disc 46 and the second part 13b of the radially inner race 13 as well as the balls 24 which innerconnect the helical grooves 23 in the first part 13a of the race 13 and the second part 13b of the race 13 although, it will be appreciated, as shown in Figures 1 and 2 a transverse section plane through disc 46 would not in practice intersect the balls 24: these are illustrated, however, in order to facilitate the explanation of the function of the bi-directional transmission of torque by the device.

Figures 3a and 3b illustrate the relative positions of the discs 46 and 47 when torque is applied clockwise, as viewed in Figure 3, from the input shaft 38 to the output shaft 18. It will be appreciated that this torque in this direction may occur in either direction of rotation of the input shaft 38. That is, assuming that the input shaft 38 is rotating clockwise, then clockwise torque will be applied if the output shaft 18 represents a load which is driven by the input shaft in the same direction. Correspondingly, if the input shaft were rotating anticlockwise and the output shaft 18 subjected to a load travelling faster than the input shaft (for example as in the over-run conditions in a motor vehicle) the torque would also be applied in a clockwise direction. In these conditions the drive shaft 38 applies torque via the radial flange 41 to the pins 44 and 45. The pins 44, 45 are thus displaced clockwise around the arcuate slots 60, 61 in the disc 47 to the ends of these slots such that there is a direct driving connection between the shaft 38 and the disc 47, and therefore via the splined coupling 53, 54, to the second shaft 21 of the first part 13a of the inner race 13. The second part 13b of the inner race 13 can, in these circumstances, turn with respect to the first part 13a as permitted by the balls 34 in the helical grooves 23, 25 to adopt an axial separation

determined by the axial separation of the radially outer race parts 14a, 14b. As shown in Figure 3b the length of the arcuate slots 62, 63 in the disc 46, together with their circumferential orientation in relation to the axial separation of the two parts 13a, 13b of the inner race 13 which, by virtue of the helical interengagement determined by the balls 24 and the helical groove 23, 25 occurs at a given relative angular orientation of the two parts 13a, 13b, is such that the pins 44, 45 are spaced from the ends of the grooves 62, 63. This configuration represents the greatest separation of the two outer race parts 14a, 14b and the closest approach of the inner race parts 13a, 13b and consequently the highest transmission ratio achievable by the transmission device 11.

Referring now to Figure 2, the lowest transmission ratio is illustrated with the two outer race parts 14a, 14b moved to their position of closest approach by rotation of the worm wheels 33, 34 as will be described in more detail below, and a consequential separation of the parts 13a, 13b of the inner race 13 allowing the planets 15 to adopt a radially inner position and determine a low transmission ratio. In this position, as in the high transmission ratio, the drive pins 44, 45 are still engaged in the ends of the slots 60, 61 of the disc 47

with drive from the input shaft 38 being transmitted, therefore, directly to the first part 13a of the inner race 13 allowing the second part 13b to move axially by relative rotation with respect to the first part 13a such that the disc 46 which is fixed for rotation therewith by the splined coupling 49, 50, 51 is turned through about 90°. It will be appreciated that the balls 50 linking the splines 49, 51 are necessary to allow this axial displacement to take place upon relative turning movement of the second part 13b of the inner race 13 with respect to the first part 13a thereof. Movement of the second part 13b axially thus applies no axial forces to the disc 46. This is an important consideration for reasons which will be explained below.

In this configuration, as illustrated in Figure 4b, the pins 44, 45 now occupy the opposite ends of the arcuate slots 62, 63 in the disc 46 from those occupied by the pins when the transmission device is in a configuration representing a high gear ratio. Again, the pins are displaced a short distance from the ends of the slots 62, 63 for reasons which will be explained hereinafter.

Between the discs 46, 47 within the annular casing 42 there is located a further disc 59 having two

diametrically opposite holes through which the pins 44, 45 pass such that the disc 59 rotates together with the flange 41 and the cylindrical sleeve 42. The parts of the connection assembly 37, namely the two discs 46, 47, intermediate disc 59 and the two casing parts comprising the flange 41 and the outer cylindrical sleeve 42 with its radial flange 43, are assembled in such a way that there is a sliding fit between the discs 46, 47 and the components in contact with their faces. This sliding fit should preferably exert a minimum frictional resistance to movement whilst having no significant play. The reason for this lies in the damping effect which can be achieved upon torque reversal as will be explained in more detail below.

Referring now to Figures 3c and 3d, these illustrate the relative positions of the discs 46 and 47 respectively with the transmission device in the high transmission ratio as illustrated in Figure 1 and the transmission of torque between the input shaft 38 and the output shaft 18 in the opposite directional sense, namely anticlockwise as viewed in the direction of the arrow A of Figure 1. In these conditions the drive pins 44, 45 are in diametrically opposite positions from those occupied in relation to the discs 46, 47 during clockwise rotation.

In this configuration the pins 44, 45 engage the ends of the slots 62, 63 in the disc 46 such that torque from the shaft 38 is now applied to the second part 13b of the inner race 13 and, because the pins 44, 45 do not travel right to the ends of the slots 60, 61 in the disc 47, the drive from this disc via the splined coupling 54, 55 to the second shaft 21 of the first part 13a of the inner race 13 is decoupled. The sense of torque transmission from the shaft 38 through the disc 46 to the second part 13b of the inner race 13 is in the same direction with respect to the first part 13a as it had been for clockwise torque transmission from the first part 13a to the second part 13b such that there is still a tendency for the first and second parts 13a, 13b to rotate with respect to one another in a sense such as to cause relative approach of these two parts thereby maintaining contact pressure between the race 13 and the planets 15.

The significance of the disc 59 between the two discs 46, 47 lies in the formation of closed chambers defined by the arcuate slots 60, 61 and 62, 63 housing the pins 44, 45 such that upon torque reversal when the slots 44, 45 move from the position illustrated in Figure 3b to that illustrated in Figure 3d the fluid trapped in the arcuate chambers defined by the slots 62, 63 and the two facing

surfaces of the members on either side of each respective disc acts to damp the movement of the pins, leaking through the interface between the adjacent contacting members as the forces continue to be applied to effect torque reversal. This damping action avoids the need for the presence of any resilient components at the ends of travel of the relatively movable parts of the coupling 37 so that, upon torque reversal, although there is a nearly 180° relative rotation between the discs 46, 47 and the shaft 38, there is no sharp impact as the direction of transmission of forces is reversed.

Figures 4a and 4b represent the relative positions of the pins 44, 45 and the slots 60, 61 and 62, 63 when the gear ratio is changed to the opposite end of the range, namely the lowermost gear ratio, in which it will be seen that the disc 46 has turned through approximately 90° to bring the slot 62 clockwise to the position in which the pin 44 is close to the opposite end of the slot from that which it occupies when the transmission device is adjusted to its highest transmission ratio. It will be appreciated that for intermediate transmission ratios the pin 44 will occupy positions within the slot 62 intermediate between these two end positions as will the pin 45 in the slot 63. Again, upon torque reversal the pins 44, 45 engage

the disc 46 as shown in Figures 4c and 4d whilst the position of these pins in the slots 60, 61 are spaced from the ends whereby to decouple the disc 47 from the first part 13a of the inner race 13.

The adjustment of the relative separation of the two parts 14a, 14b of the radially outer race 14 by rotation of the worm wheels 33, 34 is achieved by applying appropriate drive signals to the two electric motors 35, 36. It is an advantage of worm wheel drives that there is a minimum backlash such that the relative angular position of the two parts 14a, 14b and consequently their axial separation, is accurately determined. Moreover, drive cannot be transmitted from the pinion to the worm wheel due to the short pitch of the helix. Under the influence of reaction torque at the outer race assembly the action of the helical drive between the teeth 30, 31 and the worm wheels 33, 34 causes one part of the outer race 14 to react virtually all the torque (depending on the direction of torque transmission) and the other to be virtually torsionally unloaded. The loaded and unloaded members for one direction of torque transmission become the unloaded and loaded members respectively in the other direction of torque transmission thus, for either direction of torque transmission, one worm drive is free

to rotate under the action of a relatively small drive torque from the motor whilst the other will be heavily loaded due to the frictional contact. Thus, if both motors 35, 36 are energised together one will rotate easily and the other will be stalled. The relative rotation of the two motors will, however, be the same regardless of the direction of torque transmission and will be determined by the combined signal from the motor encoders. Typically, the maximum torque required to effect ratio change is 10% of the input torque and the ratio between the worm wheel and the pinions is 100:1. This produces a requirement for motor torque which amounts to 0.2% of the input torque assuming the efficiency of the worm and wheel is 50%. Ratio changes within about 0.1 seconds can be achieved by utilising a motor speed of only a few thousand RPM.

In an alternative embodiment (not shown) the two motors 35, 36 are replaced by a single motor with a differential drive transmission to the two worm drives. One advantage of utilising such worm drives, whether one or two motors are used, lies in the possibility of turning the outer race with respect to the casing and thereby causing the balls to roll even when the unit is not driven. Should it come to rest in a high ratio, therefore, the change in

configuration to a low ratio can be achieved. This is not possible with the balls stationary. This would involve, for example, driving one motor faster than the other to cause the outer race parts to move together and at the same time cause the balls 15 to turn about a radial axis as well as rolling over the races.

Referring now to Figures 7 and 8 an alternative embodiment of the invention is illustrated. Those parts which are the same as or fulfil the same function as corresponding parts in the embodiment of Figures 1 to 6 have been identified with the same reference numerals and, for simplicity, certain parts (such as the lost motion coupling 37 by which bi-directional transmission of torque is achieved) have been omitted for clarity. In this embodiment the radially inner race parts 13a, 13b are different from those in the embodiment of Figures 1 to 6 in that the curved surfaces which engage the planetary balls 15 extend radially outwardly to a point just beyond that at which the tangent to the contact surface on the planetary ball is perpendicular to the axis of rotation of the races and the planets, which is the main axis of rotation of the transmission device itself indicated by the central line in Figure 7. Thus, when the parts 14a, 14b of the radially outer race 14 are

moved apart to allow the planetary balls 15 to adopt their radially outermost position, the points of contact between the parts of the inner race 13 and the planetary balls 15 lie in respective planes perpendicular to the axis of rotation. The contact points are indicated in Figure 7 by the arrow heads A, B. Whereas, in an intermediate gear ratio, the contact points are spaced from a central axis of rotation of the planets 15 thereby causing the planet to roll in relation to the inner race 13, in this position with the contact points being aligned with the central axis of the planets 15 so that there is no radial distance between the centre of the ball and these contact points, there is no inherent tendency for the balls 15 to rotate apart from any frictional contact between the balls 15 and outer race 14 due to the outward centrifugal force applied to the balls 15 in use.

In order to decouple the balls 15 from the outer race 14 in this high ratio position the roller follower cage 17 is provided with a set of radially outwardly extending arms 65 which have pairs of "claws" 66 engaging the radially outer surface of respective balls 15. Because the roller follower cage 17 rotates with the planetary balls 15 due to the action of the roller followers 16 the

radial limitation on the position of the balls 15 by contact therewith causes no tendency for the planetary balls 13 to rotate so that they can be effectively stationary with respect to the inner race 13 once the parts 14a, 14b of the outer race 14 have been separated to move their contacting surfaces out of contact with the planetary balls 15.

In this configuration there is effectively a "direct drive" from the input shaft 38 through the coupling 37 to the second part 13b of the inner race 13 via the helical interconnection 23, 24, 25 to the balls 15 via the contact points A, B and from there to the roller followers 16, the roller follower cage 17 and the output shaft 18. This gives a nominal 1:1 ratio at the high end of the ratio range.

With the loss of contact between the planetary balls 15 and the outer race 14 the torque reaction that drives the planet balls spin also disappears and with the loss of spin, slip also disappears and the drive becomes directly coupled. Because the planetary balls 15 are now not rotating the follower rollers 16 will, likewise, not rotate in their bearings so frictional losses associated with this rotation also disappears. In this high ratio,

therefore, the spin and slip losses which are inevitable in the generation of intermediate transmission ratios are not present so that the efficiency of the drive device in high ratio is greatly increased, corresponding to the direct drive of a conventional gear box.

In an alternative embodiment (not illustrated) the radial outward movement of the planetary balls 15 is limited by the presence of an annular member rotating with these balls in place of the arms 65 and claws 66 of the follower carrier cage 17 in the embodiment of Figures 7 and 8.

In another embodiment (not shown) the magnitude of the ratio variation between the greatest and the least transmission ratio is increased by forming the inner race 13 to a smaller diameter, close to the diameter of the shaft 38 (possibly even less than this), in which case the helical interengagement may be made on a shorter portion of the race, but at a greater diameter. This allows more balls 24 to be located in the grooves than in the embodiment illustrated so that even greater axial loads can be supported.

Another refinement, not illustrated, which may be

introduced into the device accomodates the possibility that with a number of ratio changes under load the balls may tend to roll towards one end of the helical track and be unable to roll further: this would very considerably increase the frictional resistance to adjustment motion, making it difficult, if not impossible, further to change the ratios. To prevent this from occurring a coil spring may be positioned in the helical track in place of the last few balls in the set. This spring, in the shape of a coiled coil, then tends to urge the balls back down the track towards the central position each time the load on the balls is released. Such a spring may be provided at each end of the row of balls. The spring could also be used in the outer race to similar effect.

**CLAIMS**

1. A continuously variable drive transmission (11)for transmitting torque between input and output drive members (18, 38)thereof, of the type having planetary members (15)in rolling contact with radially inner and outer races (13, 14)each comprising two relatively axially movable parts (13a, 13b, 14a,14b) with control means (33,34,35,36) for selectively varying the axial separation of the two parts (14a, 14b)of one race (14)and thus the radial position of the planetary members(15) in rolling contact therewith, and means (23, 24, 25) for allowing the axial separation of the two parts (13a, 13b) of the other race (13) to vary whereby to determine the transmission ratio between the said input and output drive members (18, 38), characterised in that the two parts (13a, 13b) of the said other race (13) are helically interengaged and there are provided means (44-59) for establishing a driving connection between one of the said input and output drive members (18, 38) and one or other of the said two parts (13a, 13b) of the said other race (13) in dependence on the direction of the torque applied between them whereby to transmit torque between the said two drive members (18, 39) in either directional sense.

2. A continuously variable drive transmission according to Claim 1 , characterised in that the two parts of the said other race are helically inter-engaged with one another.

3. A continuously variable drive transmission device according to Claim 1 or Claim 2, characterised in that the drive connection to the said other race includes a lost motion coupling.

4. A continuously variable drive transmission according to any of the Claims 1 to 3, characterised in that the drive connection to the two parts of the said other race comprise two drive transmission members connected to respective parts of the said other race with a torque-direction-sensitive coupling.

5. A continuously variable drive transmission according to in any of Claims 2, 3 or 4, characterised in that the lost motion in the drive connection coupling has a maximum range greater than the range of movement of the drive connection involved in the greatest ratio change from maximum to minimum of the drive transmission.

6. A continuously variable drive transmission according to any preceding claim, characterised in that the means for interconnecting the two parts of the other race with one of the said input and output drive members comprise two rotatable drive transmission members mounted for rotation with respective parts of the said other race and driven by axially extending members passing through arcuate slots therein.

7. A continuously variable drive transmission according to Claim 6, characterised in that the said drive transmission members comprise discs mounted in a housing and dimensioned such as to be a sliding fit therein whereby to define a fluid damper chamber between the slot and the drive transmission members.

8. A continuously variable drive transmission according to any preceding claim, characterised in that the two parts of the said one race are helically inter-engaged such that relative rotation thereof causes relative axial displacement.

9. A continuously variable drive transmission according to any preceding claim, characterised in that the relative separation of the two parts of the said one race

is controlled by one or more drive motors.

10. A continuously variable drive transmission according to Claim 8 and 9, characterised in that the two said parts of the said one race are helically interengaged such that relative rotation thereof causes relative axial displacement, and the relative rotation of two parts of the said one race are controlled by one or more electric drive motors.

11. A continuously variable drive transmission according to any preceding claim, characterised in that the two parts of the said one race are located on the device such as to be capable of axial float.

12. A continuously variable drive transmission according to Claim 9, or Claim 10, characterised in that which drive motors for adjusting the relative rotation of the two parts of the said one race are linked by a worm and wheel coupling to each part.

13. A continuously variable drive transmission device (11) of the type having planetary members (15) in rolling contact with radially inner and outer races (13, 14) each associated with respective drive input and output members

(18, 38) and each comprising two axially spaced parts(13a, 13b ,14a, 14b) with control means (33, 34) for selectively varying the axial separation of the two parts (14a, 14b) of one race (14) and thus the radial position of the planetary members (15) in rolling contact therewith, means (23, 24, 25) sensitive to the torque between the said drive input and output members (38, 18) whereby to determine the separation of the two parts (13a, 13b) of the other race (13) and thus the transmission ratio of the device, characterised in that there are provided means (65) for limiting the maximum radial excursion of the planetary members (15) in one radial direction independently of the separation of the said two parts (14a, 14b) of the said one race (14), whereby to determine the transmission ratio of the device at one end of its range of adjustment.

14. A continuously variable drive transmission device according to Claim 13, characterised in that the said one radial direction is that in which the planetary members move towards the highest transmission ratio of the device.

15. A continuously variable drive transmission device according to Claim 13 or Claim, characterised in that the

drive input to the device is applied via the radially inner race and the drive output taken from a follower linked to the planetary members.

16. A continuously variable drive transmission device according to Claim 15, characterised in that the planetary members are intercalated with follower members carried on a common guide by which drive transmission from the planetary members to the output member is effected, and the common guide includes means for limiting the radially outward excursion of the planetary members.

17. A continuously variable drive transmission device according to Claim 16, characterised in that the said radial excursion limiting means comprise a part of a cage within which the planetary members are constrained to rotate.

18. A continuously variable drive transmission device according to Claim 16 or Claim 17, characterised in that the means for limiting the radial excursion of the planetary members comprises an annular member located at or adjacent the common plane of the centres of rotation of the planetary members and encircling these latter.

19. A continuously variable drive transmission device according to any of Claims 13 to 18, characterised in that the radially inner race has surfaces for contacting the planetary members which extend radially outwardly to a point such that the contact between the race and the planetary members takes place at surfaces substantially normal to the axis of rotation of the planetary members.

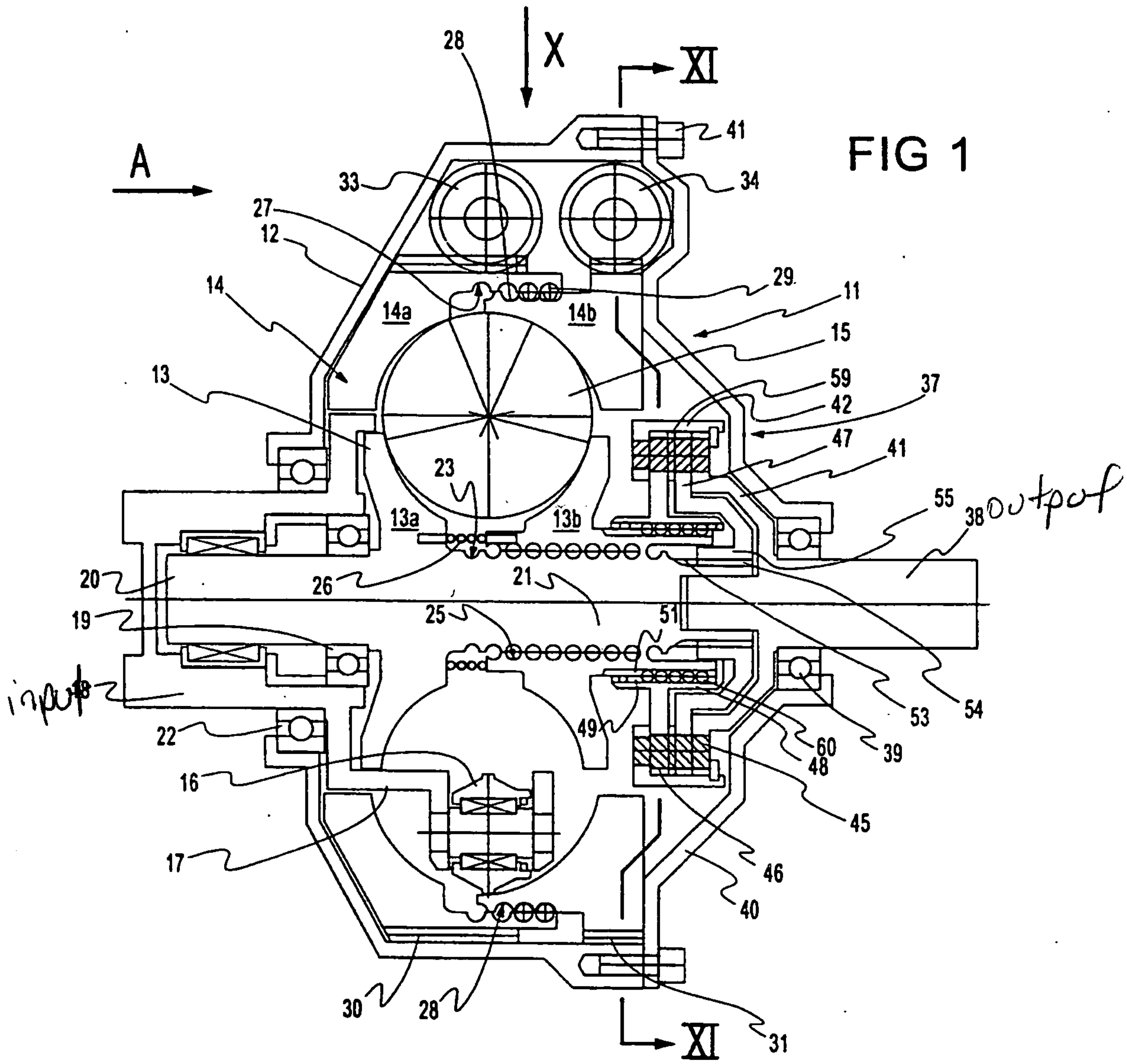

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FIG 1

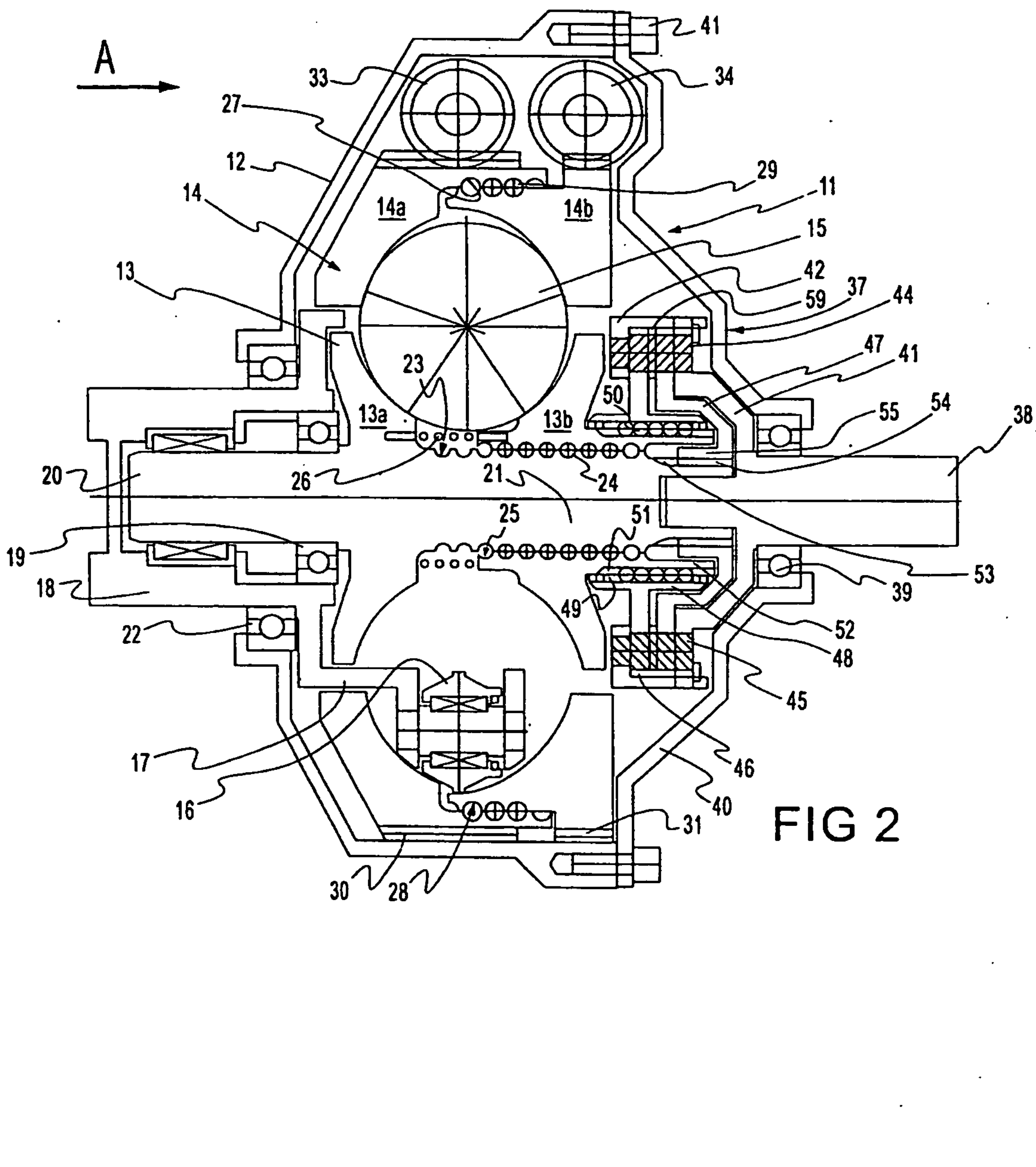
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FIG 2

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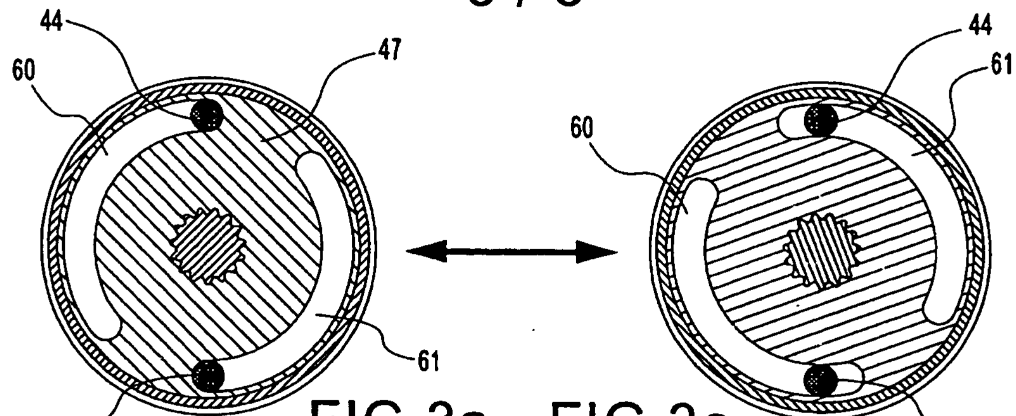


FIG 3a FIG 3c

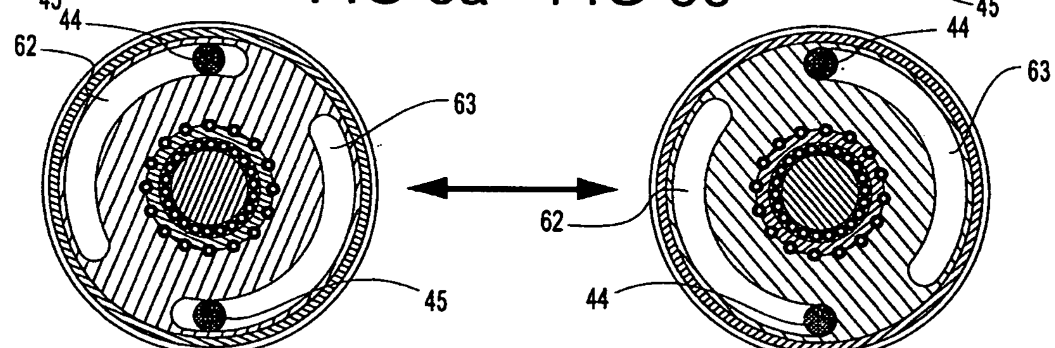


FIG 3b FIG 3d

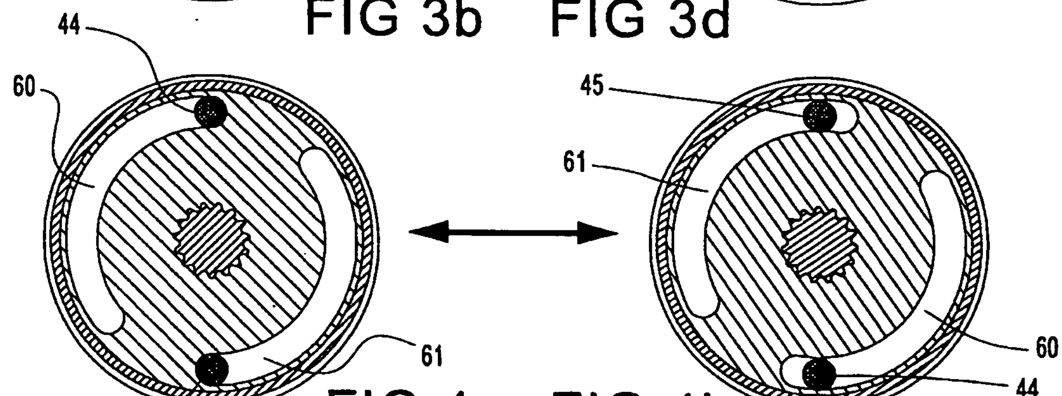


FIG 4a FIG 4b

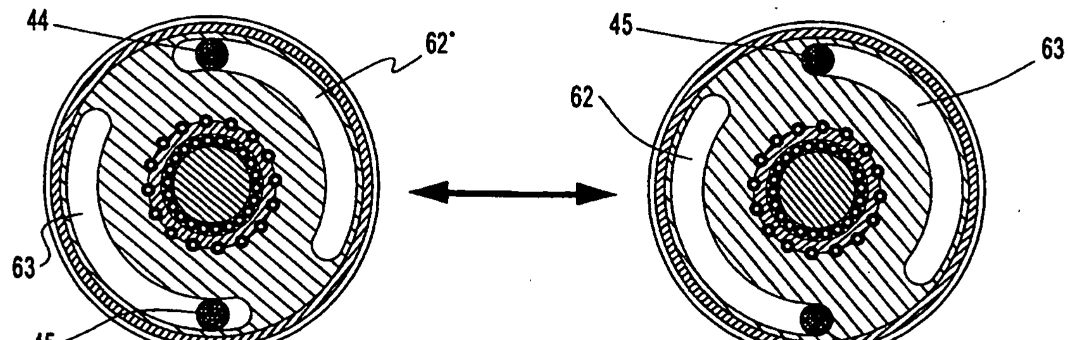


FIG 4b FIG 4d

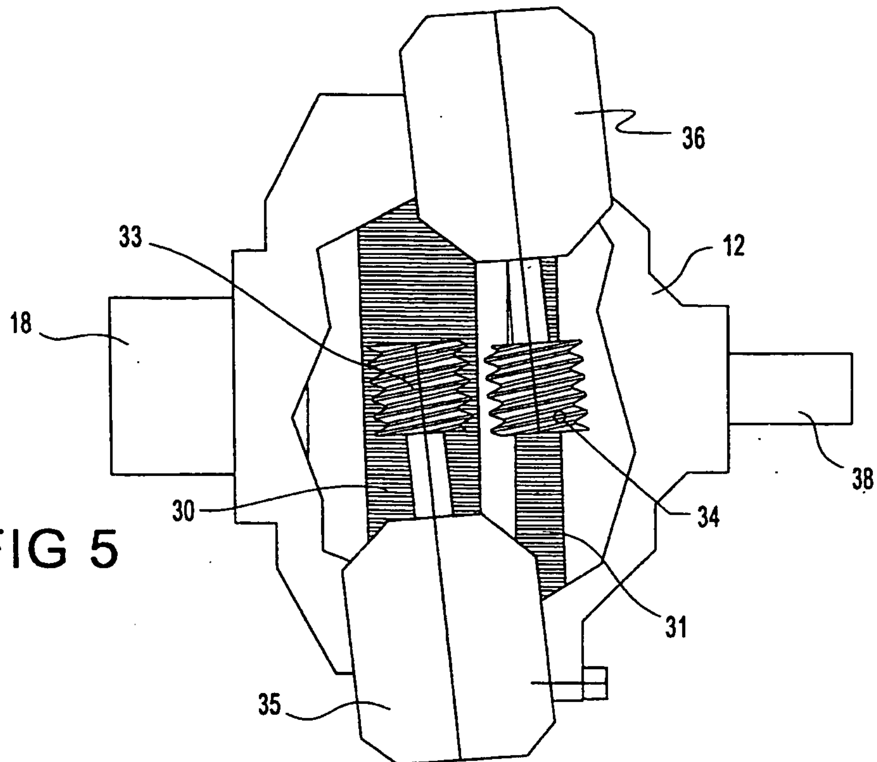


FIG 5

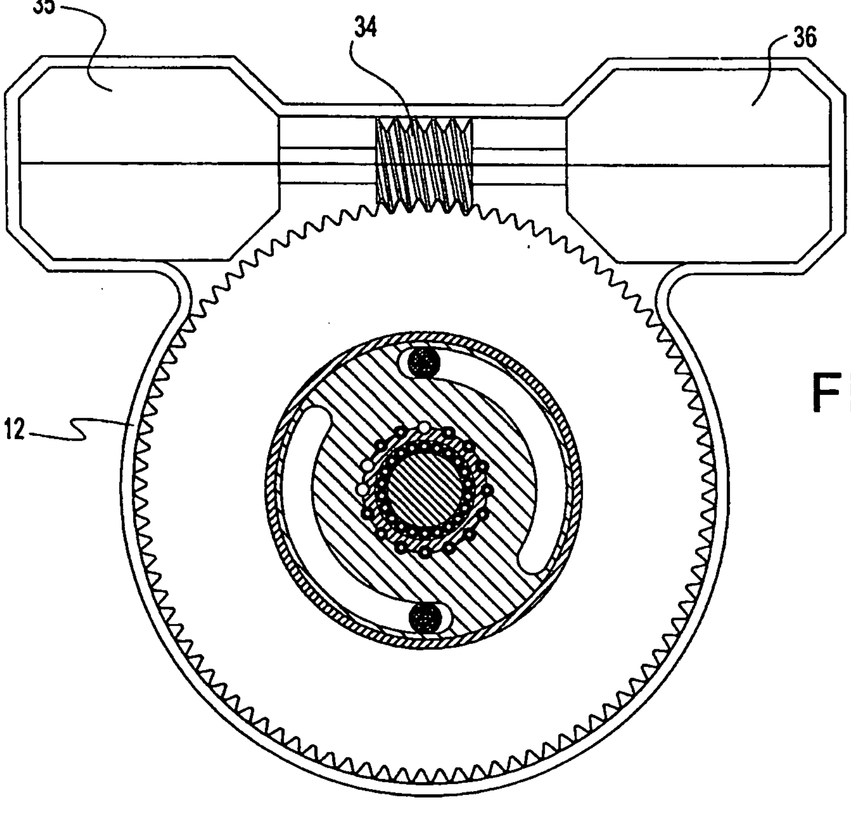


FIG 6

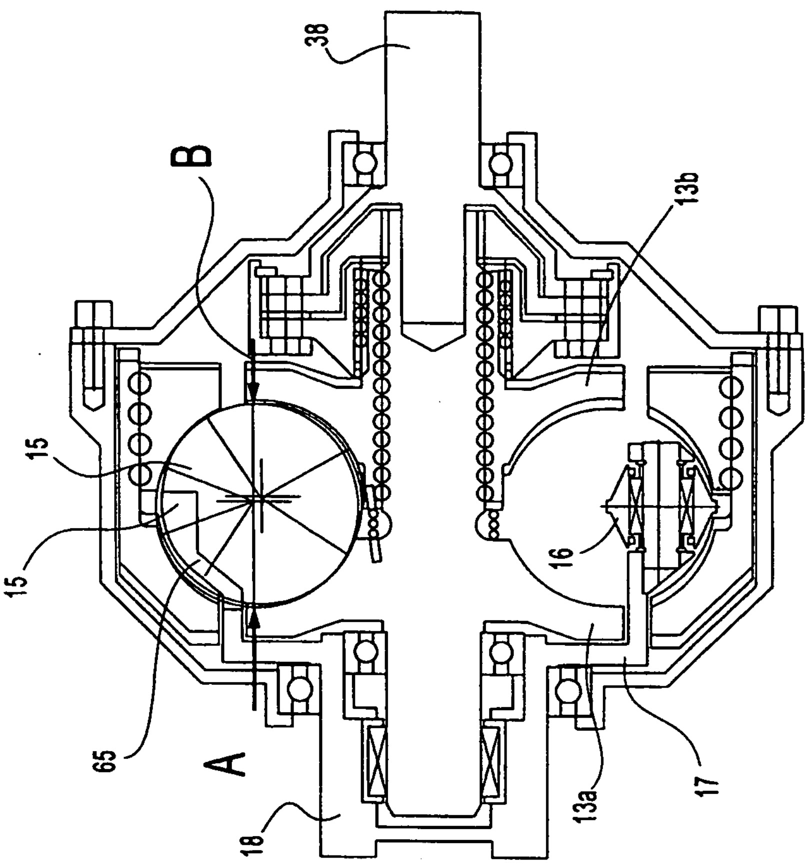


FIG 7

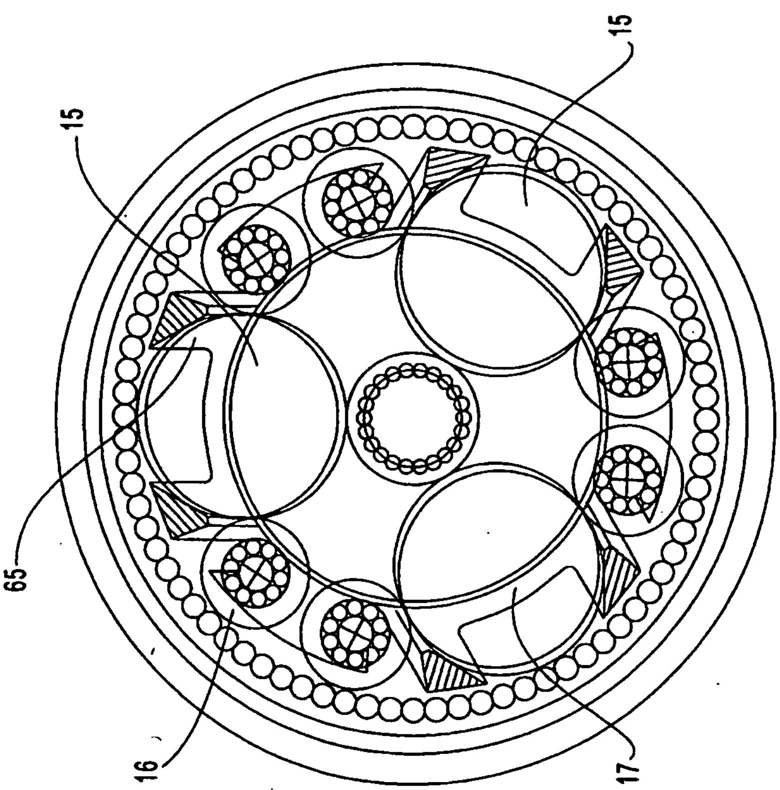


FIG 8

# INTERNATIONAL SEARCH REPORT

Internr  al Application No
PCT/GB 00/02543

**A. CLASSIFICATION OF SUBJECT MATTER**
IPC 7 F16H15/50

According to International Patent Classification (IPC) or to both national classification and IPC

**B. FIELDS SEARCHED**

Minimum documentation searched (classification system followed by classification symbols)
IPC 7 F16H

Documentation searched other than minimum documentation to the extent that such documents are included in the fields searched

Electronic data base consulted during the international search (name of data base and, where practical, search terms used)

EPO-Internal

**C. DOCUMENTS CONSIDERED TO BE RELEVANT**

| Category * | Citation of document, with indication, where appropriate, of the relevant passages | Relevant to claim No. |
|---|---|---|
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| P,Y | WO 99 35417 A (MILNER PETER J) 15 July 1999 (1999-07-15) cited in the application the whole document | 1,2 |
| P,A | | 13 |
| Y | US 4 593 574 A (SINN HARTMUT ET AL) 10 June 1986 (1986-06-10) the whole document | 1,2 |
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[X] Further documents are listed in the continuation of box C.

[X] Patent family members are listed in annex.

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"&" document member of the same patent family

| Date of the actual completion of the international search | Date of mailing of the international search report |
|---|---|
| 11 October 2000 | 19/10/2000 |

| Name and mailing address of the ISA | Authorized officer |
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| European Patent Office, P.B. 5818 Patentlaan 2<br>NL - 2280 HV Rijswijk<br>Tel. (+31-70) 340-2040, Tx. 31 651 epo nl,<br>Fax: (+31-70) 340-3016 | Van Prooijen, T |

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# INTERNATIONAL SEARCH REPORT

Intern: al Application No
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C.(Continuation) DOCUMENTS CONSIDERED TO BE RELEVANT

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Form PCT/ISA/210 (continuation of second sheet) (July 1992)

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